International Journal of Mechanical and Production Engineering Research and Development (IJMPERD)
ISSN(P): 2249-6890; ISSN(E): 2249-8001
Vol. 9, Issue 1, Feb 2019, 457-470

A TECHNICAL REVIEW ON THE STUDY OF STABILITY ANALYSIS OF NON-CIRCULAR BEARINGS LUBRICATED WITH NEWTONIAN/NON-NEWTONIAN FLUIDS

A. AMAR MURTHY[1] & RAGHUNANDANA. K[2]

[1]*Department of Mechanical and Manufacturing Engineering, Manipal Institute of Technology, MAHE, Manipal, Karnataka, India*

[2]*Department of Mechatronics Engineering, Manipal Institute of Technology, MAHE, Manipal, Karnataka, India*

ABSTRACT

In recent years, applications of non-Newtonian fluids as lubricants have received great interest since the use of these non-Newtonian lubricants showed an increase in load carrying capacity and reduction in frictional force with the escalation in additives concentration. The demands of the present day industry rotating machinery includes high speed, compactness, light weight engines, high operating loads, high power transmission, high efficiency, and high performance of the engine. As a result of high speed machines, bearings are inclined to have excessive power loss and increase in oil temperature. The former reduces the efficiency of the engine and the latter causes the undesired changes in the lubricating oil. The plain journal bearings at high rotation speeds are subjected to instability like oil whirl and whip ruining the bearing and also the machine. Therefore, this poses a need to change in the bearing design. In the present paper, an attempt is made to briefly introduce the various non-linear models used and the different approaches that have been carried out by the researches in the past few years to improve and achieve the stability of journal bearing along with enhanced performances and characteristics.

KEYWORDS: Non-Newtonian, Stability & Multilobe Bearings

Original Article

Received: Nov 28, 2018; **Accepted:** Dec 18, 2018; **Published:** Jan 23, 2019; **Paper Id.:** IJMPERDFEB201945

1. INTRODUCTION

A non-Newtonian behaviour of the fluid is observed where the stress in shear and shear strain rates are nonlinear. The incorporation of the lubricant, non-linear behaviour predicts the performance of the bearing to a more realistic and accurate results. A Newtonian fluid is mixed with various types of additives to enhance the performance characteristics like reduction in wear and friction. Normally a petroleum derived base oil and additives are commercially blended with the lubricants so as to enhance and obtain the required performance characteristics for a bearing. Generally used additive, a viscosity-index modifier, helping to prevent the uncontrollable variation in viscosity with change in temperature is used. These blending causes a change in rheology of the lubricant to non-Newtonian from Newtonian. Hence fluids can therefore be broadly classified into two categories: Newtonian and non-Newtonian. The fluid that obeys the law of viscosity and have constant viscosity are called Newtonian fluids. For a fluid that is Newtonian, the shear rate $\dot{\gamma}$ and the shear stress τ are linearly related [1] as,

$$\tau = \mu \dot{\gamma} \tag{1}$$

According to the relationships between shear stress and shear rate, non-Newtonian fluids are commonly grouped in three general classes as shown in Figure 1 and Figure 2 [2]

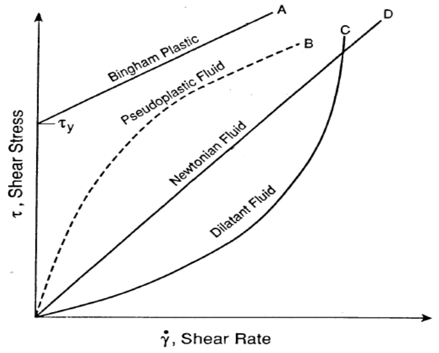


Figure 1: Typical Shear Stress [3] and Shear Rate Relationships for Non-Newtonian Fluids

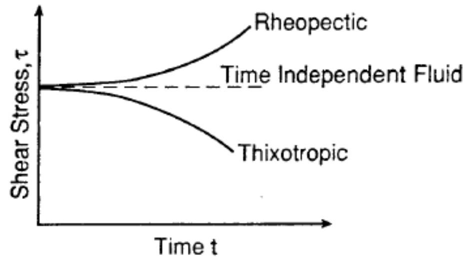


Figure 2: Time Dependent Fluids [3]

2. NON-NEWTONIAN MODELS

Out of the available mathematical models, the main four commonly used non-Newtonian fluids [4] that describe the relationship between shear rate and shear stress are:

- Power Law Model
- Bird-Carreau Model
- Cross-Power Law Model
- Herschel-Bulkley Model

3. BEARING TYPES

The demands of the present day industry rotating machinery include high speed, compactness, light weight engines, high operating loads, high power transmission, high efficiency, and high performance of the engine. As a result of high speed machines, bearings are inclined to have excessive power loss and increase in oil temperature. Different approaches have been carried out by the researches in the past few years to improve and achieve the stability of journal bearing along with enhanced performances and characteristics. Out of which a few more common and important types are discussed below.

3.1. Axial Grooving Bearings

Higher stability thresholds can be achieved in the bearings of fixed geometry by including axial grooves [5] as shown in Figure.3. The grooves provided allow the oil around the lubricated surface to flow in different pattern. More the

number of grooves that are added, the higher are the stability threshold achieved. But there is a limit for the groove addition in the bearing at which it becomes counter-productive. The load carrying capacity is effectively reduced by groove addition process thus a two groove bearing would be the better choice if load carrying capacity is the primary objective. Also these bearings can operate even when the shaft rotates in either clockwise or counter clock wise direction. For optimum flow of the lubricant, oil leakage, load carrying capacity and critical speed, the groove position is a key parameter.

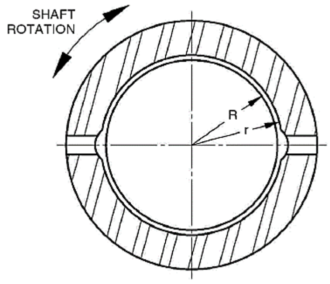


Figure 3: Two Axial Grooved Bearing [5]

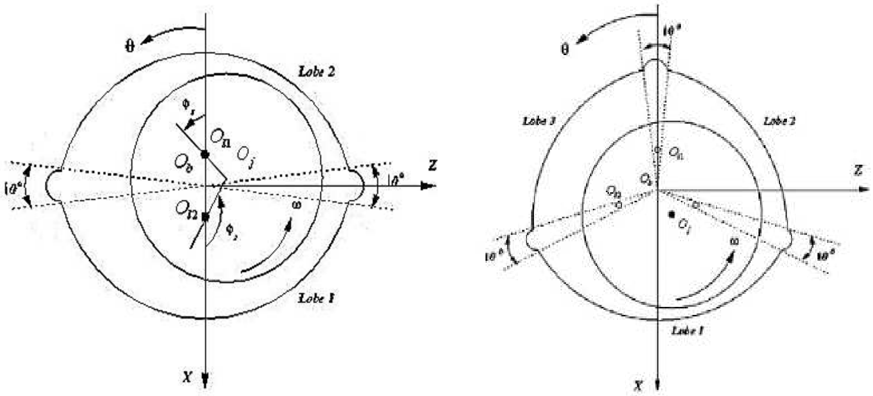
3.2 Multilobe Bearings

Rahmatabadi et al. [6] studied the steady state performance characteristics of 2-lobed, 3-lobed and four lobed bearings lubricated with micropolar fluids using finite element method. The results revealed that the static performance characteristics are enhanced significantly with the use of micropolar fluids as lubricants and are found to be prominent at larger coupling numbers. The researchers paid more attention both experimentally and analytically in the field of multilobe bearings due their specific characteristics which includes increased stability, thermal analysis as compared with the conventional circular bearings [7-14]. Said et al [15] studied dynamic stability along with the performance characteristic in steady state for 3-lobe bearing having textured surface. Their studies reveal that the bearing performance characteristics is improved to a higher extent, reduction in loss of power due to friction and improved load bearing capacity with the use of polymer thickened oils [16]. Reduction in friction and wear between the sliding surfaces is achieved with a Newtonian fluid which is blended with appropriate additives [17]. In the case of a 2-lobe and circular bearings operated with non-Newtonian fluid, a lower critical mass parameter was observed [18]. Kushare [19] studied the effect of non-Newtonian fluid used in 2-lobe hybrid bearing which was worn out and had a hole entry of symmetric type. It was found out that the non-Newtonian behavior of the fluid had an intensive effect on the bearing stability. Studied the performance of four bearings of different configurations using non-Newtonian fluid as lubricant. Das et al [20] showed that the bearing with 3-lobe has a higher load carrying capacity and lower friction variable while the bearing with 2-lobe has a higher flow coefficient. Biswas et al. [21] experimentally studied the effect of dynamic viscosity and the eccentricity ratio on a 3-lobe bearing operating at a speed of 1000 RPM. It was observed that the dynamic viscosity has a profound effect on the stability and with decrease in static loads, higher values in the stability were attained.

3.2.1 Geometry of Two-Lobe Bearing

Multilobe bearings [22-23] or profile bore bearings differ from plain cylindrical bearings in a manner that they consist of more than one separate converging region under regular operation. The centers of each bore are non-concentric and are formed with a predefined fixed offset from the shaft center. These bearings can consist of two, three or four lobes and are named accordingly. Multilobe bearings have good instability suppression as compared to plain cylindrical bearings. Figure. 4 shows few main types of non- circular profile bearings. A two lobe bearing also called as lemon bearing or

elliptical beating consist of two bores with their centers moved at a offset distance from the actual shaft center in vertical direction such that the clearance is reduced in that direction. Shims are installed during manufacturing at the horizontal split line and then a cylindrical hole is bored. Thus a lemon shape bore is obtained when the two split bore without shims are assembled.



a) 2-Lobe Bearing **b) 3-Lobe Bearing**

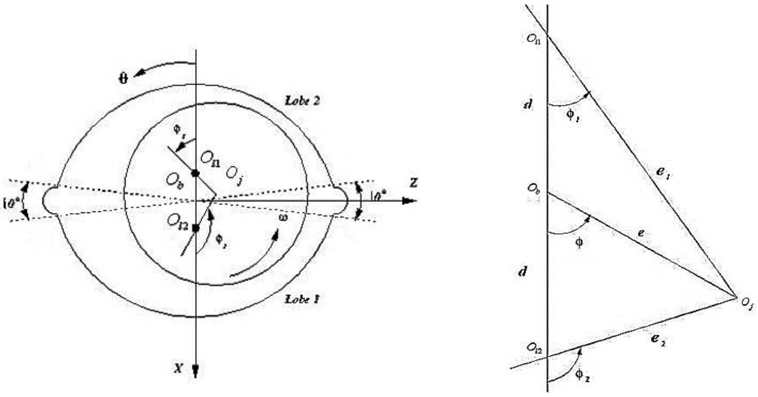
Figure 4: Lobe Bearings with Axial Grooves [22,23]

The steep convergent region in this design formed, produces preload in vertical direction on the shaft thereby pushing it towards the center hence increasing the stability. The stability of this bearing is increased when compared to the plain cylindrical bearing but with the penalties like, decrease in the load carrying capacity, increase in power loss and large amount of flow of lubricant. The lemon bearings provide adequate stability in case of bearings which are unstable marginally. The end result of profiling is mainly to provide more converging regions in the available clearance space of the bearing. The bearings in Figure. 4 (b) consisting of more than 2 lobes is generally called as multilobe bearings. These bearings provide a good suppressing of the instability in the bearing. These bearings require high flow of lubricant and most of the time will not be able to operate in shaft reverse direction. The bearing with 2-lobes is shown in the Figure.5a. The centers of each lobes are equally displaced by a distance ‘d’ from the bearing center called as ellipticity. Each of the lobes are of 170 degrees span. Axial grooves of 10 degree separates the lobes and are used for admitting the oil. In spite of enabling the oil admission in the clearance space, the grooves also helps in providing improved stability as compared to plain bearing. The relationship between attitude angle and eccentricity are acquired from Figure. 5b.

For lobe 1

$$\varepsilon_1 = \sqrt{\varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos(\phi)} \tag{2}$$

Where $\delta = \frac{d}{c}$ is the bearing ellipticity ratio and $\varepsilon_1 = \frac{e_1}{c}$ is the eccentricity ratio of lobe 1.



(a) **(b)**

Figure 5: A) 2-Lobe Bearing Geometry with Co-Ordinate System. B) Attitude Angles and Lobe Eccentricities [22,23]

Also,

$$\tan\phi_1 = \frac{e\sin\phi}{d+e\cos\phi}$$

$$\phi_1 = \tan^{-1}\frac{\varepsilon\sin\phi}{\delta+\varepsilon\cos\phi} \quad (3)$$

Similarly, for lobe 2,

$$\phi_2 = \pi - \tan^{-1}\frac{\varepsilon\sin\phi}{\delta-\varepsilon\cos\phi} \quad (4)$$

The Reynolds Equation has been derived from the Navier-Stokes equation and the continuity equation. The generalized Reynolds Equation in simplified form:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial}{\partial x}(\rho h) + 12\frac{\partial}{\partial t}(\rho h) \quad (5)$$

The film thickness for lobe 1,

$$h = c + d + e_1\cos(\emptyset - \emptyset_1 + \theta) \quad (6)$$

The film thickness for lobe 2,

$$h = c + d + e_2\cos(\emptyset - \emptyset_2 + \theta) \quad (7)$$

3.2.2. 3-Lobe Bearing Geometry

The bearing with 3-lobes [23] is shown in the Figure.6. The centers of each lobe are equally displaced by a distance ‘d’ from the bearing center called as ellipticity. Each of the lobes are of 120^0 span. Axial grooves of 10 degree separates the lobes and are used for admitting the oil. In spite of enabling the oil admission in the clearance space, the grooves also helps in providing improved stability as compared to plain bearing. The relationship between attitude angle and eccentricity are acquired from Figure. 7.

For lobe 1,

$$\varepsilon_1 = \sqrt{\varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos(\phi)} \quad (8)$$

$$\phi_1 = \tan^{-1}\frac{\varepsilon \sin\phi}{\delta+\varepsilon\cos\phi} \tag{9}$$

For lobe 2,

$$\varepsilon_2 = \sqrt{\varepsilon^2+\delta^2-2\varepsilon\delta\cos\left(\frac{\pi}{3}+\phi\right)} \tag{10}$$

$$\phi_2 = \frac{2\pi}{3} - tan^{-1}\frac{\varepsilon\sin\left(\frac{\pi}{3}+\phi\right)}{\delta-\varepsilon\cos\left(\frac{\pi}{3}+\phi\right)} \tag{11}$$

For lobe 3,

$$\varepsilon_3 = \sqrt{\varepsilon^2+\delta^2-2\varepsilon\delta\cos\left(\frac{\pi}{3}-\phi\right)} \tag{12}$$

$$\phi_3 = \frac{2\pi}{3} - tan^{-1}\frac{\varepsilon\sin\left(\frac{\pi}{3}-\phi\right)}{\delta-\varepsilon\cos\left(\frac{\pi}{3}-\phi\right)} \tag{13}$$

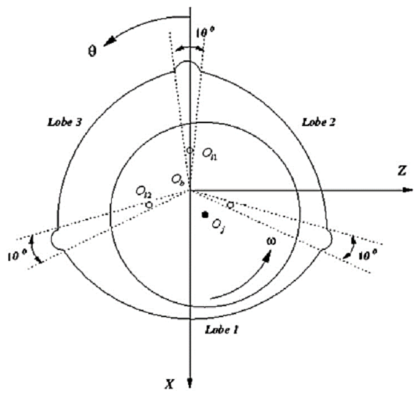


Figure 6: 3-Lobe Bearing Geometry with Co-Ordinate System.[23]

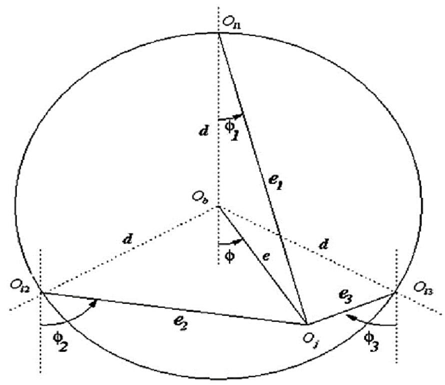


Figure 7: Attitude Angles and Lobe Eccentricities of 3-Lobe Bearing. [23]

3.2.3. Geometry of Four-Lobe Bearing

The bearing with 4-lobes [23] is shown in the Figure.8. The centers of each lobes are equally displaced by a distance ‘d’ from the bearing center called as ellipticity. Each of the lobes are of 90^0span. Axial grooves of 10 degree separates the lobes and are used for admitting the oil. In spite of enabling the oil admission in the clearance space, the grooves also help in providing improved stability as compared to plain bearing. The relationship between attitude angle and eccentricity are acquired from Figure. 9.

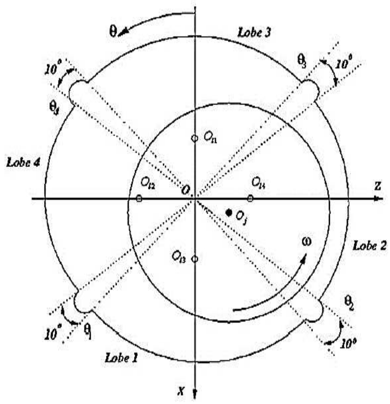


Figure 8: 4-Lobe Bearing Geometry with Co-Ordinate System.

For lobe 1

$$\varepsilon_1 = \sqrt{\varepsilon^2 + \delta^2 + 2\varepsilon\delta\cos(\phi)} \tag{14}$$

$$\phi_1 = \tan^{-1}\frac{\varepsilon \sin\phi}{\delta + \varepsilon\cos\phi} \tag{15}$$

For lobe 2

$$\varepsilon_2 = \sqrt{\varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos\left(\frac{\pi}{2} + \phi\right)} \tag{16}$$

$$\phi_2 = \frac{\pi}{2} - tan^{-1}\frac{\varepsilon\, sin\left(\frac{\pi}{2}+\phi\right)}{\delta - \varepsilon\, cos\left(\frac{\pi}{2}+\phi\right)} \tag{17}$$

For lobe 3

$$\varepsilon_3 = \sqrt{\varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos(\phi)} \tag{18}$$

$$\phi_2 = \pi - tan^{-1}\frac{\varepsilon\, sin(\phi)}{\delta - \varepsilon\, cos(\phi)} \tag{19}$$

For lobe 4

$$\varepsilon_4 = \sqrt{\varepsilon^2 + \delta^2 - 2\varepsilon\delta\cos\left(\frac{\pi}{2} - \phi\right)} \tag{20}$$

$$\phi_2 = \frac{\pi}{2} - tan^{-1}\frac{\varepsilon\, cos(\phi)}{\delta - \varepsilon\, \sin(\phi)} \tag{21}$$

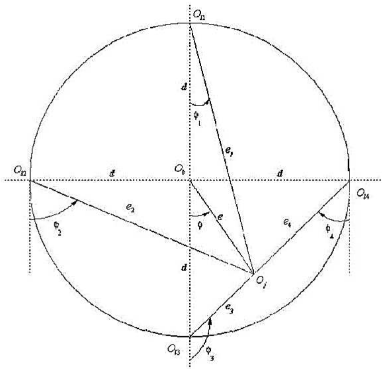


Figure 9: Attitude Angles and Lobe Eccentricities of 4-Lobe Bearing [23]

3.3. Pressure Dam Bearings [24]

These bearings as shown in Figure. 10 are nothing but a slight modified form of plain axial groove bearing with a relief track or a dam cut mostly in the top bore of the bearing. This bearing is considered as the more stable in the available fixed geometry bearing types. High pressure is developed in these dams imposing more forces on the shafts and making it appear heavier thereby forcing it to position in a more stable condition. The power consumption of these bearings is high and they are usually expensive. These bearings are effective when the dams are properly designed with the idea of its angular position, width, depth and magnitude of the load acting along with its direction. The load carrying capacity of pressure dam bearing is high. These bearings operate in unidirectional only. If an ordinary plain bearing has to be designed for improvement in terms of stability without replacement, then milling a pressure dam is an ideal choice.

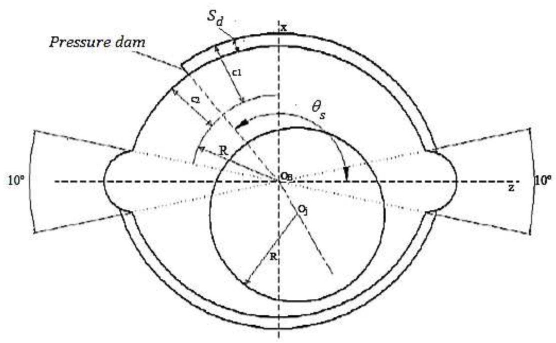


Figure 10: Pressure Dam Bearings with Grooves

3.4. Offset Half Bearings

The offset half bearing shown in Figure. 11 is also a unidirectional configuration of bearing. This bearing is same as plain bearing but the top half is horizontally shifted resulting in two converging regions. The load bearing capacity of these bearings are increased with the increase in the offset.

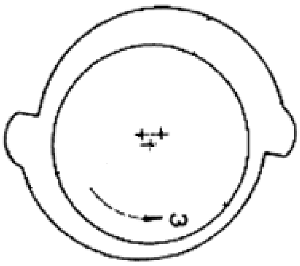


Figure 11: Offset Half Bearings

4. STABILITY OF BEARINGS

The sub-synchronous whirl stability limit can be determined by two methods: Nonlinear transient method and linearized perturbation method. The velocity and displacement of oil film are used to determine the effect of hydrodynamic forces. Usually resources [25-27] prefer a non –linear instability oil-whirl analysis approach instead of a linear type analysis. As the perturbation technique is simpler and involves less computation requirements, this method is the most widely used methods. A more understanding of the post-whirl nature and more correct results can be obtained by nonlinear transient analysis. Philip [28] provides a detail description of the most commonly used non-circular bearings. Rahmani et al. [29] explored the stability of bearings along with the static performance characteristics like load carrying capacity, friction coefficient in terms of non-linearity and ellipticity parameters along with the non-Newtonian fluid. The results showed that there is a decrease in the load carrying capacity and increase in friction coefficient but increase in stability in comparison with that of a normal plain circular bearing using Newtonian fluid. Akkok et al. [30] experimentally studied and gave the results for load carrying capacity and whirl onset in bearings of different shapes like circular, offset halves and elliptical bearing. They confirmed by experiments the validity of the linearized model for calculating whirl. Experiments conducted by them showed that increase in groove size may have a destabilizing effect rather than the positive effect of preloading. Also it was found that the stability characteristics of offset halves bearings were best when they are lightly loaded while the elliptical bearings are superior at heavy load conditions. Rao et. al. [31] studied the effect of different L/D ratios and ramp size on the dynamic performance of a tri-taper bearing. The stability of a rigid rotor supported on a tri-taper journal bearing, subjected to periodic and variable rotating loads was predicted using a nonlinear transient analysis. The results obtained revealed that the increase in ramp size resulted in good stability of the bearing under constant unidirectional loading and variable rotating load but an increase in L/D ratios with same loading conditions resulted in larger excursions of journal shaft. It was also found that the journal is subjected to instability for a periodic load. Another interesting point to be observed was that the region of stability increased with an increase in length to diameter ratio for a given ramp size and a similar result was achieved for a given length to diameter ratio with an increase in ramp size. Rattan et al. [32] studied the effect of the orientation of the load on the stability of the three lobes bearing with pressure dam. The analysis was carried out for a bearing that supports a rigid or a flexible rotor. The results showed that the stability of the bearings is increased when the load line is shifted by some degrees in the direction opposite to the shaft rotation for both rigid and flexible rotor. It was also found that with the increase in the flexibility of the rotor, the infinite stability zone was unaffected but the minimum value of the threshold speed was decreased there by reducing the stability of the bearing. Nair et al[33] carried out an analysis using micro polar lubricant in an elliptical bearing providing the details of the effect of deformation of the bearing liner on the dynamic and static characteristics of a bearing. Raghunandana et al. [34] investigated the effect of non-Newtonian lubricant on of performance characteristic of plain journal bearing. A modified Reynolds equation was devel-

oped using Dien and Elrod model for the non-Newtonian model. Stability analysis was carried out using a non-linear transient method. The studies showed that with the use of non-Newtonian fluid, the stability of the bearing was improved. Mishra [35] numerically studied the effect of misalignment of the journal, non-circularity of bore, along with non-Newtonian lubricant behaviour. The output obtained were the temperature, thermal pressure, stiffness, damping factor, non-dimensional critical mass and whirl ratio. Stability of the bearing was influenced due to the collective effect of non-circularity, eccentricity, misalignment and roughness of surface. It was found that the bearing with isotropic pattern roughness showed better stability. It was found that stability was related directly to non-dimensional critical mass and was related inversely to whirl ratio. Nabarun Biswas and K.M. Pandey [36] studied the three lobed bearing with lobe having a span of 1200. The shaft rotation speed was taken as 60000 rpm. It was observed that the performance of the bearing was greatly affected with the presence of lobes. It was concluded that the providing lobes in the bearing would definitely enhance the bearing's life span. Chiang et al. [37] studied the effect of couple stress along with the roughness of the surface on the instability threshold of a rotor bearings. Using the linear theory, the coefficients of damping and stiffness were determined. The results showed an increase in stiffness and the damping coefficients ensuing in a greater stability threshold speed. While in the transverse roughness case there was a decrease in the threshold speed as compared with the smooth surfaced bearing using couple stress lubricant. The longitudinally roughed surface lubricated with couple stress fluid resulted in the reduced attitude angle and the steady eccentricity ratio. Also for a small disturbance, the longitudinally roughed bearings were more stable as compares to the smoother ones. Bhushan et al. [38] analysed the dynamic behaviour of four-lobe pressure-dam bearing. It was produced by incorporating two pressure dams in the upper two lobes and two relief tracks in the lower two lobes of an ordinary four-lobe bearing. The characteristic curves of this bearing showed that it was much more stable as compared to an ordinary four-lobe bearing. It was observed that with the incorporation of pressure dams and relief tracks, the stability of a four-lobe bearing increased. The geometric parameters that affected the stability of the four-lobe pressure-dam bearing were dam depth, dam location, dam width and relief-track axial width. The optimized values of these parameters were determined. It was found that with increase in relief-track axial width, the stability of the four-lobe pressure-dam bearing increased and also the stability increased with decrease in L/D ratio. Chetti [39] carried out a numerical analysis to study the performance characteristics of a 4-lobed bearing operated with couple stress fluid in both static and dynamic state. A modified Reynolds equation was modelled and was solved using finite difference method. The studies showed that, with the increase in the couple stress parameter, the load carrying capacity was increased and there was a decrease in friction coefficient. Also the bearing was found to be more stable. Mehta [40] studies revealed that the load carrying capacity of a two lobe bearing operated with couple stress fluid was increased even when the bearing was operated at lower values of eccentricity ratio.

5. CONCLUSIONS

From the literature studies carried out, the following conclusions can be drawn.

- The use of additives to enhance the lubricant properties causes it to behave as non-Newtonian and this in turn affects the performance characteristic of the bearing.
- The use of non-Newtonian lubricant in the multi-lobe bearings, offset halves, axial groove bearings etc. has a positive effect on the bearing performance like reduction in friction, good thermal characteristics, higher damping and stiffness coefficients, better load carrying capacity and increased stability.
- The performance characteristic of lobed bearings is higher when compared to that of a plain circular bearing.

- Non-circular bearings like axial groove bearing, offset half, pressure dam bearings have higher stability as compared to that of plain circular bearings.

REFERENCES

1. *Bird, R. B., W. E. Stewart, and E. N. Lightfoot. Transport Phenomena, Wiley, New York, 1960.*
2. *Skelland, Anthony Harold Peter. Non-Newtonian flow and heat transfer. Wiley, 1967.*
3. *Wu, Yu-Shu. Theoretical studies of non-Newtonian and Newtonian fluid flow through porous media. No. LBL-28642. Lawrence Berkeley Lab., CA (USA), 1990.*
4. *Open FOAM v5 User Guide: 7.3 Transport/rheology models. Retrieved from https:// cfd. direct / openfoam / user-guide / transport - rheology.*
5. *Stachowiak, Gwidon, and Andrew W. Batchelor. Engineering tribology. Butterworth-Heinemann, 2013.*
6. *Malik M, Sinhasan R, Chandra M. Design data for three lobe bearing. ASLE Trans 1981; 24:345–53.*
7. *Rahmatabadi AD, Rashidi R. Effect of mount angle on static and dynamic characteristics of gas-lubricated, noncircular journal bearings. IJST 2006;37:27–37.*
8. *Taylor DV, Kostrzewsky GJ, Flack RD, Barrett LE. Measured performance of ahighly preloaded three-lobe journal bearing-part I: static characteristics. Tribol Trans 1995; 38:507–16.*
9. *Nair KP, Nair VPS, Jayadas NH. Static and dynamic analysis of elastohydrodynamic elliptical journal bearing with micropolar lubricant. TribolInt2007;40:297–305.*
10. *van Ostayen RAJ, van Beek A. Thermal modelling of the lemon-bore hydrodynamic bearing. TribolInt2009;42:23–32.*
11. *Mehta NP, Rattan SS, Bhushan G. Static and dynamic characteristics of fourlobe pressure-dam bearings. Tribol Lett 2003; 15:415–20.*
12. *Shukla, D. K., SINGH, M., & Jain, K. K. (2014). Bearing Capacity Of Footing On Slopping Anisotropic Rock Mass. IMPACT: International Journal of Research in Engineering & Technology (IMPACT: IJRET), 2(6), 217-232.*
13. *Ghoneam SM, Strzelecki S. Thermal problems of multilobe journal bearing tribosystem. Meccanica 2006; 41:571–9.*
14. *Ma MT, Taylor CM. A comparative thermal analysis of the static performance of different fixed profile bore plain bearings. Proc Inst Mech Eng J JEngTribol 1999; 213:13–30.*
15. *Crosby WA, Chetti B. The static and dynamic characteristics of a two-lobe journal bearing lubricated with couple-stress fluid. Tribol Trans 2009;52:262–268.*
16. *EL-Said, A. K., EL-Souhily, B. M., Crosby, W. A., & EL-Gamal, H. A. (2017). The performance and stability of three-lobe journal bearing textured with micro protrusions. Alexandria Engineering Journal, 56(4), 423-432.*
17. *Oliver DR. Load enhancement effects due to polymer thickening in a short model journal bearing. J Non-Newton Fluid Mech 1988; 30:185–9.*
18. *Scott W, Suntiwattana P. Effect of oil additives on the performance of a wet friction clutch material. Wear 1995; 181–183:850–5.*
19. *Sinhasan, R., &Goyal, K. C. (1995). Transient response of a two-lobe journal bearing lubricated with non-Newtonian lubricant. Tribology international, 28(4), 233-239.*

20. *Kushare, P. B., & Sharma, S. C. (2014). Nonlinear transient stability study of two lobe symmetric hole entry worn hybrid journal bearing operating with non-Newtonian lubricant. Tribology International, 69, 84-101.*

21. *Das, B. J., & Roy, L. (2018). Analysis and comparison of steady-state performance characteristics of two-axial groove and multilobe hydrodynamic bearings lubricated with non-Newtonian fluids. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology,2018*

22. *Biswas, N., Chakraborti, P., Saha, A., & Biswas, S. (2016, July). Performance & stability analysis of a three lobe journal bearing with varying parameters: Experiments and analysis. In AIP Conference Proceedings (Vol. 1754, No. 1, p. 030002). AIP Publishing.*

23. *Roy, Lintu, and S. K. Kakoty. Groove location for optimum performance of three-and four-lobe bearings using genetic algorithm. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 2015, 229.1, 47-63.*

24. *Roy L, Kakoty SK, Groove Location for Optimum Performance of Two-Lobe Bearing Using Genetic Algorithm. International J Swarm Intel Evol Compu2rt. 2015 4:120.*

25. *Ghosh, S. Inertia Effect under Couple Stress Fluid In Laminar Flow Condition In Journal Bearing.*

26. *Roy, L., and A. Choudhury. Dam Length Optimization of Two Lobe Pressure Dam Bearing Using Genetic Algorithm. J App. Mech. Engg., 2014, 3.145 2.*

27. *Majumdar, Bankim C., David E. Brewe, and Michael M. Khonsari. Stability of a rigid rotor supported on flexible oil journal bearings. Journal of tribology, 1988, 110.1, 181-187.*

28. *Turaga, Ram, A. S. Sekhar, and B. C. Majumdar. Comparison between linear and nonlinear transient analysis techniques to find the stability of a rigid rotor." Journal of tribology, 1999,121.1, 198-201.*

29. *Majumdar, Bankim C and David E. Brewe. Stability of a rigid rotor supported on oil-film journal bearings under dynamic load. No. NASA-E-3727. National Aeronautics and Space Administration Cleveland Oh Lewis Research Center, 1987.*

30. *Kiameh, Philip. Power generation handbook: selection, applications, operation, and maintenance. McGraw-Hill Professional, 2003.*

31. *F. Rahmani, R. K. Pandey, and J. K. Dutt. Influence of Elliptic Bore and non-Newtonian Rheology of Lubricant on the Performance and Stability of Journal Bearing.*

32. *Akkok M, McC, Ettles C M. The onset of Whirl instability in journal bearings of various bore shapes and groove sizes, Journal of lubrication technology, 1983, 105, 342-352.*

33. *D S Rao, B S Shenoy, R S Pai, R Pai, Stability of tri-taper journal bearings under dynamic load using a non-linear transient method, Tribology International, 2010, 9, 43, 1584-1591.*

34. *S S Rattan, N P Mehta, G Bhushan. Effect of load orientation on the stability of a three-lobe pressure dam bearing with rigid and flexible rotors, Journal of Engineering and Technology, 2011,1, 10.*

35. *K. Prabhakaran Nair, V. P. Sukumaran Nair and N. H. Jayadas, "Static and dynamic of elastohydrodynamic elliptical journal bearing with micropolar lubricant, " Trib. Inter., vol. 40, pp. 297-305,2007.*

36. *Raghunandana K, Jhon, Kanthraj A N. Stability of Journal Bearings Considering Slip Condition: A Non Linear Transient Analysis, International Journal of Applied Science & Technology Research Excellence, 2013, 1, 3, 46-50.*

37. *Prakash Chandra Mishra. Mathematical modeling of stability in rough elliptic bore misaligned journal bearing considering thermal and non-Newtonian effects, Applied Mathematical Modelling, 2013, 8, 37, 5896-5912.*

38. *N Biswas, K M Pandey. Transient CFD Analysis of Multi-Lobe Bearings at 60000 RPM for A Gas Turbine, International Journal of Engineering, 2011, 5, 3, 512.*

39. *Chiang HL, Lin JR, Hsu CH, Chang YP, Linear stability analysis of a rough short journal bearing lubricated with non-Newtonian fluids, Tribology Letters, 2004, 4, 7, 867-877.*

40. *Bhushan G, Rattan S S, Mehta NP. Stability analysis of four-lobe pressure-dam bearings, Tribology Letters, 2002, 1, 13, 1-7.*

41. *Chetti, B. Static and Dynamic Characteristics Of Hydrodynamic Four-Lobe Journal Bearing With Couple Stress Lubricants. Jordanian International Mechanical Engineering Conference (JIMEC'7). 2010.*

42. *Mehta, N. P., Rattan, S. S., and Verma, R. Stability analysis of two lobe hydrodynamic journal bearing with couple stress lubricant. ARPN Journal of Engg. And Applied Sciences, 2010, 5(1), 69-74.*